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Specification

Double-row antifriction bearing

The invention relates to a double-row antifriction bearing which has a one-piece bearing ring and a bearing ring divided into two parts in the axial direction, and rolling elements located between them, the rolling elements of the first row of rolling elements having a first diameter and the rolling elements of the second row of rolling elements having a second diameter which is different from the diameter of the rolling elements of the first row of rolling elements.

An antifriction bearing of the generic type is described for example in GB 152 018. Bearings of this type are used for example in high performance motor vehicles, i.e. in auto racing. In particular, the gear shafts of a race car are supported with these bearings. Since the requirements for rpm and stiffness of the bearing are very high here, the bearings are moreover made as four-point bearings; but this limits the forces which can be transmitted. At correspondingly high loads and large rpm known antifriction bearings quickly reach their limits.

Therefore the object of the invention is to develop an antifriction bearing of the initially mentioned type such that it is suited to accommodating higher forces at very high rpm, as occur especially in the transmissions of race cars.

The inventions's achieving this object is characterized in that the rolling elements of the two rows of rolling elements consist of ceramic material and the contact angle of one row of rolling elements is different from the contact angle of the other row of rolling elements.

Preferably the one-piece bearing ring is the outer ring of the bearing and the split bearing ring is the inner ring.

A value in the range between 5° and 35° has proven itself as the contact angle of one row of rolling elements. Conversely, preferably the contact angle of the other row of rolling elements in the range between 10° and 60° is used.

As known in the indicated applications in auto racing, the outer ring preferably has a flange molded on in one piece. This flange can be located with respect to its axial position at the height of one of the rows of rolling elements.

For optimum supply of the bearing with lubricant it is provided according to the development that lubrication openings are made in the contact area of the front surfaces of the split bearing ring. Furthermore it can be provided that the outer ring is provided with lubrication openings, especially with lubrication holes.

Preferably the rows of rolling elements have cages which are guided on one shoulder of at least one of the bearing rings. Advantageously the cages are guided on one shoulder of the split bearing ring. Plastic, preferably PEEK, has proven itself as the material of the cage.

The rolling elements are preferably made as balls. The bearing thus assumes the design of an angular contact bearing.

The proposed antifriction bearing is preferably used as a component of a transmission which in operation has very high rpm and a high temperature, especially in a race car.

With the proposal as claimed in the invention an antifriction bearing is devised which under extreme conditions has good running behavior and a relatively long service life. It is optimally suited to use in the transmission of a race car in which very high forces occur at high rpm and high operating temperatures.

One exemplary embodiment of the invention is shown in the drawings.

Figure 1 shows a radial section through a double-row antifriction bearing (section A-B

according to Figure 2) and

Figure 2 shows a side view of this bearing.

The antifriction bearing 1 shown in the figures has a one-piece bearing ring, the outer ring 2, and a split bearing ring, specifically the inner ring 3. Between the outer ring 2 and the (split) inner ring 3 there are rolling elements 4 and 5 in the form of balls of ceramic material. Ceramic balls in an antifriction bearing are known as such in the prior art. The rolling elements 4 and 5 each form a row 6 and 7 of rolling elements, respectively.

The antifriction bearing 1 here is made as an angular contact bearing. The contact area between the rolling elements 4, 5 and their respective raceways in the bearing rings 2, 3 is therefore at an angle to the radial direction. The contact angles are labelled a_1 and a_2 for the two rows 6 and 7 of rolling elements, respectively. The contact angle a_1 of the first row 6 of rolling elements is in the range between 5^o and 35^o. Conversely the contact angle a_2 of the second row 7 of rolling elements is in the range between 10^o and 60^o.

Furthermore, it can be seen that for the two rows 6 and 7 of rolling elements balls 4, 5 with different diameters d_1 and d_2 are used. The diameter d_1 of the first row 6 of rolling elements is much smaller than the diameter d_2 of the second row 7 of rolling elements; it is preferably in the range between 50% and 80% of the diameter d_2.

The pitch circles of the two rows 6 and 7 of rolling elements are different. The one of the first row 6 of rolling elements is smaller than that of the second row 7 of rolling elements. Preferably the radius of the pitch circle of the first row 6 of rolling elements is between 85% and 95% of the radius of the pitch circle of the second row 7 of rolling elements.

As furthermore follows from the figures, the outer ring 2 is made such that its radially outer periphery passes into a flange 8 which is provided with a number of through holes with

which the outer ring 2 can be fixed on a machine part. The axial position of the flange 8 is located in this connection exactly at the height of the first row 6 of rolling elements.

The two inner rings 3 are opposite one another on the facing side with their front surfaces 9. It is suggested that there is a lubricating opening 10 here with which it is possible to supply the contact point between the rolling elements 4, 5 and the raceways with lubricant.

The two rows of rolling elements 6, 7 in the known manner also have cages 11. They consist of plastic, preferably of the plastic material PEEK. The two inner rings 3 each have one shoulder 12 against which the cage 11 rests and is guided.

The osculations between the balls 4 and 5 and their raceways in the bearing rings 2, 3 are conventionally selected such that low-friction running results. The same applies to the choice of the bearing slackness for the two rows of rolling elements.